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# NUMERICAL INVESTIGATION OF COPPER PIPE GROUP'S VIBRATION LEVEL IN AN AIR CONDITIONER

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## ABSTRACT

Air conditioners performs four basic tasks such as cooling, heating and humidity control to provide the desired comfort in their environment. While providing all these, it is important that it does not increase the ambient noise as another comfort parameter for the consumers. Therefore, noise and vibration reduction on air conditioners is a demanding task. In split type air conditioners, the main sound source of indoor unit is fan and air flow. For outdoor unit, sound sources are much more: fan, compressor and structural vibration. In this study, the displacement of the copper pipe group in the outdoor unit of a split type air conditioner is determined by modal analysis method. Then, the numerical model is improved and verified by comparing the obtained results with the experimental results. Finally, pipes that do not meet the design criteria in the reference design are modified and system vibration is reduced with the verified model.

**Keywords:** *Air conditioner, modal analysis, single rotary compressor, finite element method*

## 1. INTRODUCTION

Split type air conditioners have wide range of usage area like home and small shops to control inside temperature and humidity. The performance of these units is crucially dependent on factors such as cooling capacity, air flow rate, and energy consumption. Additionally, consumer comfort is influenced by vibration and sound levels. The main vibration source is compressor that is located in outdoor units. This produced vibration transmitted along the copper pipe group [1]. At some point, vibration with high amplitude may cause damage on pipes. In such cases cooling performance of the air conditioner may get worse because of refrigerant leakage [2]. Even in cases where the pipes remain undamaged, high vibration levels can result in noise issues. This structural noise can be disturbing to the end user. Therefore, a thorough analysis is required to ensure the proper design and compatibility of the pipe group with the compressor.

This study aims to replace the compressor in an existing outdoor unit and examine the compatibility of the new compressor with the copper pipe group, as well as determine the requirements for the new pipe design. Initially, a numerical model of the pipe group with the new compressor was created using NX NASTRAN to investigate its vibration behavior. Subsequently, areas of the pipe group with significant displacement were identified. Finally, design proposals were developed to address the identified displacement areas and improve the overall design.

## 2. INITIAL SYSTEM INVESTIGATION

System components may undergo changes in order to enhance system performance, simplify production processes, or improve cost-effectiveness. In the context of this study, the twin rotary compressor in the outdoor unit was replaced with a single rotary compressor. The specifications of these two compressor are outlined in Table 1. Total weight of the single rotary compressor and its charge limit are 100 gr and 320 gr, respectively which are lighter than twin compressor. Additionally, the maximum rotational speeds differ, with the single rotary compressor operating at 7200 rpm, while the twin compressor reaches a maximum of 7800 rpm. These variations contribute to differences in system vibration levels.

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In order to ensure cooling system with the new rotary compressor matches with the vibration design criteria, it will be investigated with computational methods. New system need be improved until there is no pipe displacement over 0.7 mm.

**Table 1.** Compressor specifications.

| | Twin comp. | Single comp. |
|---|---|---|
| Refrigerant charge | 1020 g | 700 g |
| Revolution range | 10 -130 rps | 12 – 120 rps |
| Displacement | 10.2 $cm^3$/rev | 8.92 $cm^3$/rev |
| Oil charging amount | 280 ml | 280 ml |
| Weight | 6.4 kg | 6.3 kg |
| Refrigerant | R32 | R32 & R410a |
| Vibration | 7 $m/s^2$ | 15 $m/s^2$ |
| Rated capacity | 3912 W | 2785 W |

In an air conditioner outdoor unit there are many vibration sources like compressor, refrigerant flow in the pipes, outdoor unit fan motor and air flow induced vibration. Since it is complex to include all these effects, and this study focus on just compressor change; vibration level investigation of the system will be based on structural vibration of compressor and nearby pipes. Therefore, compressor will be considered as the only excitation mechanism. After the compressor replacement, the existing piping system has been directly connected to the new compressor. To test the structural compatibility of this new system, a numerical model has been created in Siemens NX environment.

The numerical study first started with model division into small cells. In the model, 1461 beam elements in 1D, 50689 CQUAD(4) elements in 2D and 6932 CTETRA(10) tetra elements in 3D, total of 59,082 elements were created. Figure 1 indicates the used cell types for different part of the system and fixed constraints. Then, structural analysis was performed using the NX Nastran solver with SOL 103 Response Dynamics solution type.

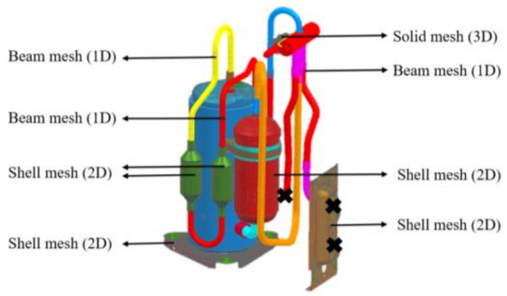



**Figure 1.** Cell types of the system model.

The numerical modal analyses conducted on the numerical model of the single rotary compressor and pipe assembly revealed the presence of specific modes in different regions. In the vicinity of the silencers, the first and second modes were observed at frequencies ranging from 16 to 18 Hz. Around the four-way valve, the third and fourth modes were identified at frequencies between 27 and 39 Hz. Additionally, the fifth mode was observed at 54 Hz, while the sixth mode occurred at 61 Hz.

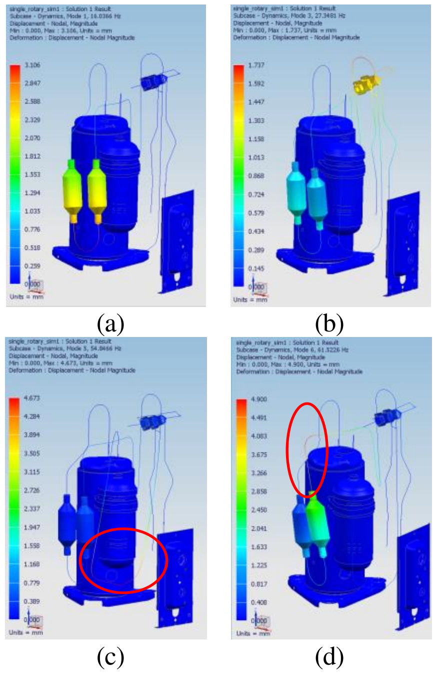



**Figure 2:** (a) 1st and 2nd modes, (b) 3rd and 4th modes, (c) 5th mode, (d) 6th mode of the system.

To validate the computational analysis of the pipe group, a vibration experiment was conducted. For the experimental study Brüel & Kjaer Pulse LabShop program was used with Pulse analyzer, B&K 4394 single axes accelerometer. Vibration data was collected while the compressor was operating within the frequency range of 23 Hz to 83 Hz. Based on the experimental measurements, the highest displacements were observed in the frequency range of 43 Hz to 61 Hz at the pipe bend section where accelerometer number 4 was positioned. This comparison between the computational analysis and experimental measurements helps to verify the accuracy and reliability of the numerical model in capturing the actual vibration behavior of the pipe group. It also highlights the critical frequency range and specific location that requires attention for further analysis and design improvements.

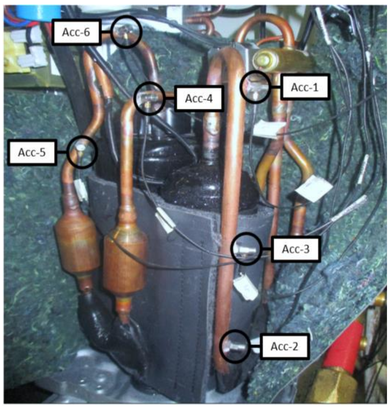



**Figure 3:** Accelerometer placement on the copper pipe group.

Figure 4 shows the displacement levels obtained from six accelerometer locations. It can be observed that there are more vibrations at the locations of accelerometer number 1, 2 and 4.

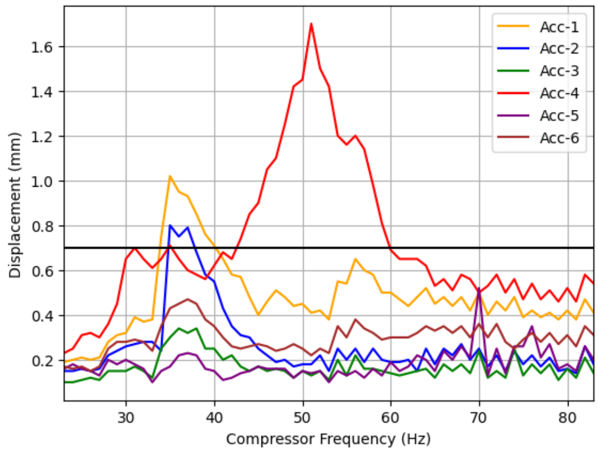


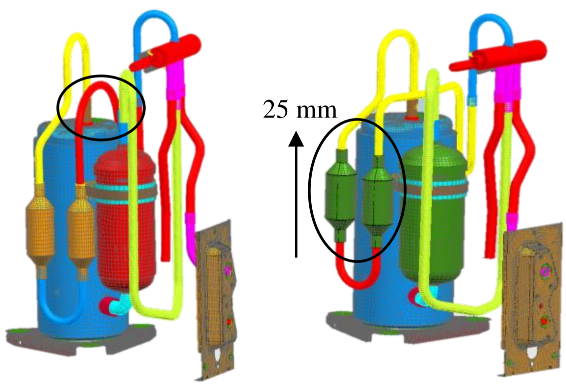

**Figure 4:** Pipe displacement through different compressor speed.

The modal frequency results obtained from the experimental measurements were found to be consistent with those obtained from the numerical modal analysis. However, nonlinear components such as grommets in the physical product cannot be fully modeled in the NX Nastran program. Therefore, even though the modal frequencies in the experimental measurements and numerical modal analysis match completely, there may be slight differences in vibration amplitudes. There is a ratio of 0.35 between the experimental measurement and numerical modal analysis. Therefore, the displacement values obtained from numerical modal analyses will be multiplied by this ratio and compared with the results of experimental measurements.

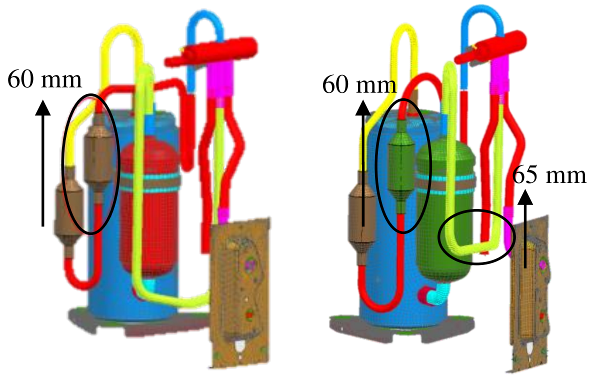
## 3. IMPROVEMENT IN DESIGN

In order to meet with 0.7 mm displacement limitation criteria, design improvements were carried out at the 5th mode at 54 Hz and the 6th mode at 61 Hz according to the results of experimental measurements and numerical modal analyses. During the design optimization studies, the pipe modes in the frequency range of 10 Hz – 80 Hz were shifted above the operating frequency. To achieve this, the modes of pipes with modal frequencies close to 80 Hz were shifted beyond the operating frequency with the help of the designed modifications. The bend radius of the pipe was also taken into consideration during the design process. Different design studies were carried out within the constraints to reduce the vibrations of pipes that could not be shifted to higher modal frequencies.

The first improvement in design focuses on the accelerometer 4 location. As shown in Figure 5-a, bend radius of the black circled area is increased. According to the modal analysis results of this design, the vibrations that were observed at a level of 1.7 mm in the experimental measurements were reduced to 1.55 mm.




(a) (b)




(c) (d)

**Figure 5:** (a) Improved design 1, (b) Improved design 2, (c) Improved design 3, (d) Improved design 4.

In another proposed design shown in Figure 5-b, two silencers have been shifted 25 mm upwards. Due to the considerably long length of the pipe that exits the silencer and enters the four-way valve group, high amplitude vibrations are observed in that section. By shifting the silencer 25 mm upwards, the length of the pipe is shortened and the vibration of the pipe is reduced. The displacement amplitude reduced from 1.7 mm to 1.5 mm with this design change. In the design proposal shown in Figure 5-b, it was observed that moving the silencer up by 25 mm and reducing the length of the pipe decreased the pipe vibrations. Therefore, in the new design proposal shown in Figure 5-c, the silencer was moved up by 60 mm from its current position to further reduce the pipe vibrations, resulting in a shorter pipe. According to the modal analysis conducted for this design, the displacement value observed in experimental measurements, which was 1.7 mm, decreased to 1.35 mm.

Figure 5-d shows the new design numbered 4. In this design, the yellow pipe where high displacement vibrations were observed at 61 Hz, as measured by accelerometer number 2, was moved 65 mm upward. As a result, the pipe length was shortened compared to the original design. Numerical analysis shows that the mode at 61 Hz has shifted to a frequency of 82 Hz. Although the pipe's vibrations did not decrease with the shortening of the pipe, the mode frequency has been shifted outside the compressor operating frequency range of 10 – 80 Hz. Therefore, there will be no resonance frequency since the system's operating frequency and the pipe's mode will not coincide.

Design number 5 with a 4-coil spring pipe with a 20 mm radius is shown in Figure 6. This design resulted in a reduction of the displacement levels seen in experimental measurements from 1.7 mm to the range of 0.6 - 0.7 mm at the location where accelerometer number 4 was placed, according to the frequency response function results obtained (Figure 7). These vibration levels were obtained without placing any vibration-damping material on the pipes.

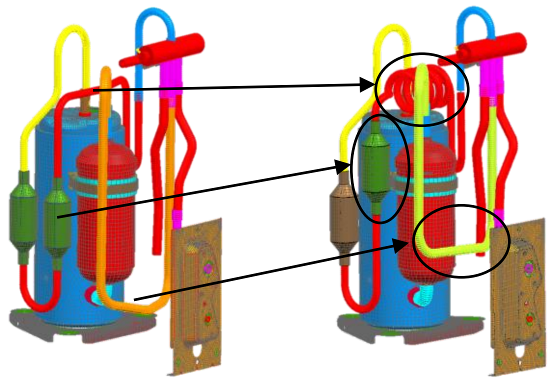


**Figure 6:** Improved design 5.

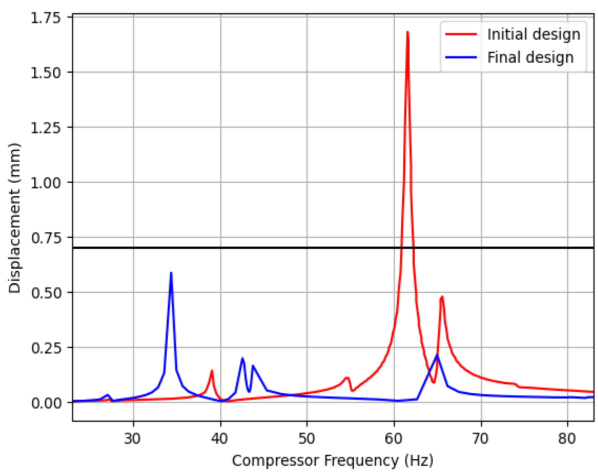



**Figure 7:** Pipe group displacement of the initial design and improved design 5.

## 4. CONCLUSION

In the present paper, vibration levels of the copper pipe group in an air conditioner outdoor unit with a replaced compressor were investigated using numerical methods. Firstly, a finite element model of the existing pipe group was established, and the natural frequencies and operating frequencies of the system were examined. Additionally, the vibration and displacement levels of the pipes were investigated. The obtained results were compared with experimental displacement data, and the relationship with numerical data was established. After a detailed analysis of the existing model, problematic regions and frequencies were identified, and design proposals targeting these regions were developed. Considering the displacement levels, it was revealed that the 5th design proposal kept the pipe fatigue/failure limit below 0.7 mm, as desired.

## 5. ACKNOWLEDGMENTS

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